Analysis of the stresses in the corner zone of a C-frame press

Citation for published version (APA):

Document status and date:
Published: 01/01/1979

Document Version:
Publisher’s PDF, also known as Version of Record (includes final page, issue and volume numbers)

Please check the document version of this publication:

• A submitted manuscript is the version of the article upon submission and before peer-review. There can be important differences between the submitted version and the official published version of record. People interested in the research are advised to contact the author for the final version of the publication, or visit the DOI to the publisher’s website.
• The final author version and the galley proof are versions of the publication after peer review.
• The final published version features the final layout of the paper including the volume, issue and page numbers.

Link to publication

General rights
Copyright and moral rights for the publications made accessible in the public portal are retained by the authors and/or other copyright owners and it is a condition of accessing publications that users recognise and abide by the legal requirements associated with these rights.

• Users may download and print one copy of any publication from the public portal for the purpose of private study or research.
• You may not further distribute the material or use it for any profit-making activity or commercial gain
• You may freely distribute the URL identifying the publication in the public portal.

If the publication is distributed under the terms of Article 25fa of the Dutch Copyright Act, indicated by the “Taverne” license above, please follow below link for the End User Agreement:
www.tue.nl/taverne

Take down policy
If you believe that this document breaches copyright please contact us at:
openaccess@tue.nl
providing details and we will investigate your claim.

Download date: 03. Aug. 2019
Analysis of the stresses in the corner zone of a C-frame press.

U.P. Singh, Eindhoven University of Technology/NL; P.C. Veenstra, Eindhoven University of Technology/NL (1); J.A.H. Ramaekers, Eindhoven University of Technology/NL.

Summary: It is observed in practice that the corner zone of a press-frame is highly sensitive to rupture-failure. The extant classical methods to evaluate the stressed state of the corner zone appear to be inadequate to predict such a failure. The application of both the finite element method and the finite difference method proves to be laborious and rather expensive. By means of the application of the theory of the thin walled-curved beam with large curvature it is possible to analyse the stresses in a corner zone with fair reliability. In the present paper this method is applied to the corner zone of a mechanical C-frame-press.
1. INTRODUCTION

Both experimental and theoretical works conducted by several investigators [1, 2, 3] confirm that the corner zone of a press frame is the most heavily stressed section. The actual stresses in this section are considerably higher than those calculated by conventional methods of calculation.

The application of the finite element method using beam elements in analysing the stress-deformed state of a corner zone does not provide results with adequate accuracy to meet the present demand of modern press designs. Although the application of the finite difference method as well as the finite element method using more sophisticated elements such as rectangular, tetrahedronal or rectangular-hexahedronal etc. can provide more accurate results [4, 2], the use of these methods is restricted in general usage due to cost of labour and computation. Hence simpler approaches for reasonably reliable estimation of the stresses at corner zones are needed.

In view of the practical application an attempt is made in the present paper to analyse the stresses at the corner zone of 20 x 10^4 kN-C frame press (fig. 1.a) by thin walled-curved beam theory and compared with other methods like the classical straight beam (conventional) and the finite beam element. Based on both the theoretical analysis and experiments, it is
shown that this approach viz., the theory of the thin wall-curved beam with large curvalure provides results which agree better with practice. It will be shown that the influence of deformation of the contour of cross-sectional profile is accounted for if the actual flange width (2a) is replaced in such a way that the elementary theory of bending applied to such a transformed section (equivalent cross section) gives the correct value of maximum bending stress [5].

Fig. 1.a. Schematic view of $20 \times 10^4$ kN C-frame press.
1. b. Cross-sectional view of the corner zone.

2. ANALYSIS

1. c. Equivalent cross-section of the corner zone.
2.1. Formulation of the problem

For the study of the corner zone the cross-section as mentioned in fig. 1.c. and subjected to concentrated load is relevant. Basically the following is assumed:

1. When circular plates (web and ribs) are subjected to symmetrical loading, they may be analysed as per theory, referred to in [6].

2. Since the condition $R_2 \geq R_1 + (H_1 + H_2)/2$ holds (as shown in fig. 2), the shape of the outside edge has virtually no influence on the stiffness [7].

3. Since the thickness of the cylindrical plate (flange) is much smaller than the overall height of the cross-section, the situation is one of pure bending for analysing the plate in the t-direction (as shown in fig. 4.b).

![Fig. 2. The shape of the outside edge of an angular element according to Kaminskaya.](image)

With the above mentioned assumptions the object of the study would be:

a. to obtain an equivalent cross-section from the equilibrium conditions of the coupling at the junction between the flanges and the web,

b. to determine the stresses in the flanges from the theory of cylindrical shells,

c. to determine the stresses in the web from the theory of symmetrical bending of circular plates and

d. finally to apply the results so obtained to analyse the stressed-deformed state of the corner zone.
2.2. The model and evaluation

According to Panamarov-Biderman [8] for the case of pure bending, the circular plate (web) is considered to be loaded through the cylindrical plates (flanges) which in turn are assumed to be cylindrical shells as shown in fig. 3.

\[ R = \text{the radius of curvature.} \]
\[ h = \text{thickness of the flange.} \]
\[ n = \text{distance between the central layer of the flange and the neutral axis of the cross-section.} \]

Fig. 3.a. Spatial view of the I-sectional profile.
Fig. 3.b. View of cylindrical plate (flange) element before deformation.
Fig. 3.c. View of flange element after deformation

Under the action of an external load a local displacement \( w \) of a cylindrical plate occurs in radial direction. The strain of a cylindrical plate in tangential direction is written as:

\[
\varepsilon_t = \frac{n + z}{R + z} \cdot \frac{\Delta \theta}{\theta} - \frac{w}{R + z} \tag{1}
\]

Since the thickness \( h \) of the cylindrical plate compared to \( n \) and \( R \) is small, the equation (1) can be written as:

\[
\varepsilon_t = \frac{n}{R} \cdot \frac{\Delta \theta}{\theta} - \frac{w}{R} \tag{2}
\]

The strain of the cylindrical element in axial direction is represented by:
\[ \varepsilon_x = \frac{K'L' - KL}{KL} = \varepsilon_o + z \frac{d^2w}{dx^2} \]  

(3)

The axial and tangential stresses in the cross-section of the cylindrical element are determined as:

\[ \sigma_x = \frac{E}{1-\nu} \left( \varepsilon_x + \nu \varepsilon_t \right) = \frac{E}{1-\nu} \left[ \varepsilon_o + z \frac{d^2w}{dx^2} + \nu \left( \frac{n}{R} \cdot \Delta \frac{\partial \theta}{\partial x} - \frac{w}{R} \right) \right] \]  

(4)

\[ \sigma_t = \frac{E}{1-\nu} \left( \varepsilon_t + \nu \varepsilon_x \right) = \frac{E}{1-\nu} \left[ \frac{n}{R} \cdot \Delta \frac{\partial \theta}{\partial x} - \frac{w}{R} + \nu \varepsilon_o + \nu z \frac{d^2w}{dx^2} \right] \]  

(5)

where \( \varepsilon_o \) is the strain in the central layer.

The forces per unit length on the cylindrical element are found to be:

\[ T_x = \int_{-h/2}^{h/2} \sigma_x dz = \frac{Eh}{2} \left[ \varepsilon_o + \nu \left( \frac{n}{R} \cdot \Delta \frac{\partial \theta}{\partial x} - \frac{w}{R} \right) \right] \]  

(6)

\[ T_t = \int_{-h/2}^{h/2} \sigma_t dz = \frac{Eh}{2} \left[ \frac{n}{R} \cdot \Delta \frac{\partial \theta}{\partial x} - \frac{w}{R} - \nu \varepsilon_o \right] \]  

(7)

and the corresponding moments per unit length are:

\[ M_x = \int_{-h/2}^{h/2} \sigma_x zdz = D \frac{d^2w}{dx^2} \]  

(8)

\[ M_t = \int_{-h/2}^{h/2} \sigma_t zdz = \nu D \frac{d^2w}{dx^2} \]  

(9)

where \( D = \frac{Eh^2}{12(1-\nu^2)} \) is the flexural rigidity of a flange element.

E and \( \nu \) are the modulus of elasticity and the poisson ratio of the material of the element, respectively.

When considering the equilibrium of the element as shown in fig. 4.b. with respect to forces along x and z-axes and the moments about t-axis, the following is obtained

\[ \varepsilon_o = -\nu \left( \frac{n}{R} \cdot \Delta \frac{\partial \theta}{\partial x} - \frac{w}{R} \right) \]  

(10)

\[ T_t = Eh \left( \frac{n}{R} \cdot \Delta \frac{\partial \theta}{\partial x} - \frac{w}{R} \right) \]  

(11)

\[ \sigma_x = \frac{E}{1-\nu} \cdot \frac{d^2w}{dx^2} z = \frac{12 M_x}{h} \frac{z}{h^3} \]  

(12)

\[ \sigma_t = \frac{T_t}{h} + \frac{12 M_t}{h^3} \frac{z}{h} \]  

(13)

\[ \frac{d^2M_x}{dx^2} = \frac{1}{R} T_t \]  

(14)
Fig. 4.a. Deformation of the cylindrical plate element.

Fig. 4.b. Equilibrium condition of the plate element.
Substituting the expressions (8) and (11) for $M_x$ and $T_t$ into equation (14) results in:

$$\frac{Eh^3}{12(1-\nu^2)} \cdot \frac{d^4w}{dx^4} + \frac{1}{R} Eh \left( \frac{n}{R} \cdot \frac{\Delta d\theta}{d\theta} - \frac{w}{R} \right)$$

(15)

The expression in equation (15) can be written in the form of a differential equation which gives the relative deformation of the cylindrical plate as follows:

$$\frac{d^4w}{dx^4} + 4 \ k \ w = 4 \ k^4 \ \eta \ \frac{\Delta d\theta}{d\theta}$$

(16)

where

$$k = \frac{4 \sqrt{3(1-\nu^2)}}{R h^2}$$

(17)

The general solution of the equation (16) is given as [8]:

$$w = \eta \ \frac{\Delta d\theta}{d\theta} + C_1 V_1(kx) + C_2 V_2(kx) + C_3 V_3(kx) + C_4 V_4(kx)$$

(18)

where $C_1$, $C_2$, $C_3$ and $C_4$ - constants of integration,

$V_1$, $V_2$, $V_3$ and $V_4$ - functions of Krilov which are defined as:

$$V_1(kx) = \text{ch}(kx) \ \cos (kx);$$

$$V_2(kx) = \frac{1}{2} \ [\text{ch}(kx) \ \sin (kx) + \text{sh}(kx) \ \cos (kx)];$$

$$V_3(kx) = \frac{1}{2} \ \text{sh}(kx) \ \sin (kx);$$

$$V_4(kx) = \frac{1}{4} \ [\text{ch}(kx) \ \sin (kx) - \text{sh}(kx) \ \cos (kx)]$$

(19)

2.3. Application of the model

As a definition for the equivalent flange width it is introduced:

$$2a_{\text{eq}} = K 2a$$

Where $K$ and $2a$ are the width reduction factor and the actual flange width respectively.

2.3.1. The boundary conditions

By setting the origin of the coordinates at the periphery of the inside flange (point 0 in fig. 1.b) the following boundary conditions are obtained:

For $x = 0$: 
\[
\begin{align*}
(Mx_1)_{x=0} &= \left( \frac{d^2 w_1}{dx^2} \right)_{x=0} = 0; \\
(Qx_1)_{x=0} &= \left( \frac{d^3 w_1}{dx^3} \right)_{x=0} = 0; \\
(Mx_2)_{x=0} &= \left( \frac{d^2 w_2}{dx^2} \right)_{x=0} = 0; \\
(Qx_2)_{x=0} &= \left( \frac{d^3 w_2}{dx^3} \right)_{x=0} = 0
\end{align*}
\]

\((20)\)

For \( x = a_1,2 \):

\[
\begin{align*}
(w_1)_{x=a_1} &= 0; \\
\left( \frac{dw_1}{dx} \right)_{x=a_1} &= 0; \\
(w_2)_{x=a_2} &= 0; \\
\left( \frac{dw_2}{dx} \right)_{x=a_2} &= 0
\end{align*}
\]

\((21)\)

2.3.2. Solution

From equations (20) and (21) it follows:

\[ C_{31} = C_{32} = C_{41} = C_{42} = 0 \]

Hence the expression (18) for displacements of outside and inside flanges can be written as:

\[
w_1 = n_1 \frac{\Delta \theta}{d\theta} 
\left[ 1 - \frac{V_1(k_1 a_1) V_2(k_1 x) + 4V_4(k_1 a_1) V_2(k_1 x)}{V_1^2(k_1 a_1) + 4V_4(k_1 a_1) V_2(k_1 x)} \right] \]

\[ w_2 = n_2 \frac{\Delta \theta}{d\theta} 
\left[ 1 - \frac{V_1(k_2 a_2) V_2(k_2 x) + 4V_4(k_2 a_2) V_2(k_2 x)}{V_1^2(k_2 a_2) + 4V_4(k_2 a_2) V_2(k_2 x)} \right] \]

\((22)\)

The forces per unit length on the inside and outside flanges respectively are determined as:

\[
T_{t_1} = Eh_1 \left( \frac{n_1}{R_1}, \frac{\Delta \theta}{d\theta} - \frac{w_1}{R_1} \right); \\
T_{t_2} = Eh_2 \left( \frac{n_2}{R_2}, \frac{\Delta \theta}{d\theta} - \frac{w_2}{R_2} \right)
\]

\((23)\)

The total forces acting on the inside and outside flanges respectively are:

\[
F_{t_1} = \int_0^{2a_1} T_{t_1} \, dx = E \frac{\Delta \theta}{R_1 d\theta} n_1 h_1 2K_1 a_1; \\
F_{t_2} = \int_0^{2a_2} T_{t_2} \, dx = E \frac{\Delta \theta}{R_2 d\theta} n_2 h_2 2K_2 a_2
\]

\((24)\)
where

\[ K_1 = \frac{1}{k_1 a_1} \frac{\text{sh} \left( 2k_1 a_1 \right) + \sin \left( 2k_1 a_1 \right)}{2 + \text{ch} \left( 2k_1 a_1 \right) + \cos \left( 2k_1 a_1 \right)}; \]

\[ K_2 = \frac{1}{k_2 a_2} \frac{\text{sh} \left( 2k_2 a_2 \right) + \sin \left( 2k_2 a_2 \right)}{2 + \text{ch} \left( 2k_2 a_2 \right) + \cos \left( 2k_2 a_2 \right)} \]

where \( K_1 \) and \( K_2 \) are the width reduction factors of the inside flange and outside flange respectively.

The total forces acting on the web and the ribs respectively are determined as:

\[ F_{t1}^* = h_3 E \frac{\Delta \theta}{d\theta} \left( b_n z dz \right) = h_3 E \frac{\Delta \theta}{d\theta} \left( b_n \ln \frac{R_2}{R_1} \right); \]

\[ F_{t2}^* = 2h_4 E \frac{\Delta \theta}{d\theta} \left( b_n z dz \right) = 2h_4 E \frac{\Delta \theta}{d\theta} \left( b_n \ln \frac{R_1}{R} \right) \]

b and \( h_3 \) are the height and thickness of the web respectively, whereas \( b_n \) and \( h_4 \) are those for the rib.

Since the normal force on the cross-section, in case of pure bending, is absent, it holds:

\[ F_{t1} + F_{t2} + F_{t1}^* + F_{t2}^* = E \frac{\Delta \theta}{d\theta} \left[ \sum \frac{n_1}{R_1} h_1 2K_1 a_1 + \frac{n_2}{R_2} h_2 2K_2 a_2 + \right. \]

\[ \left. + h_3 \left( b_n \ln \frac{R_2}{R_1} \right) + 2h_4 \left( b_n \ln \frac{R_1}{R} \right) \right] = 0 \]

The equation (27) allows to determine the radius of curvature of the neutral axis as follows:

\[ \rho_n = \frac{h_1 2K_1 a_1 + h_2 2K_2 a_2 + b h_2 + 2b h_4}{R_1} \]

\[ + \frac{h_3 \ln \frac{R_2}{R_1} + 2h_4 \ln \frac{R_1}{R}}{R} \]

The bending moment acting on the cross-section is determined as:

\[ M = F_{t1} n_1 + F_{t2} n_2 + h_3 \oint_{1} \sigma_{t1}^* z dz + \oint_{2} \sigma_{t2}^* z dz \]

where \( \sigma_{t1}^* \) and \( \sigma_{t2}^* \) are the stresses in the web and the rib respectively.
Substituting equations (24) and $\sigma_\ell^*_t = \frac{z}{\rho_\ell^z} E \frac{\Delta d\theta}{d\theta}$, into equation (28) results in:

$$M = E \frac{\Delta d\theta}{d\theta} \left[ \frac{n_1^2}{R_1} h_1 2K_1 a_1 + \frac{n_2^2}{R_2} h_2 2K_2 a_2 + h_3 b \left\{ \frac{b}{2} - \eta_1 + \rho_n \left( \frac{\rho_n}{b} - 1 \right) \right\} + 2h_4 b_1 \left\{ \frac{\rho_n}{b_1} - \frac{b_1}{2} - \eta_1 \right\} \right]$$

(30)

For a curved beam with large curvature it holds:

$$M = E \frac{\Delta d\theta}{d\theta} S$$

where \( S = \frac{1}{R} \), the modified static moment of the cross-section with respect to the neutral axis.

Analogous to this, for a curved beam with large curvature when taking into account the deformation of cross-section contour it is proposed:

$$M = E \frac{\Delta d\theta}{d\theta} S^{eq}$$

(31)

From equations (30) and (31) it follows:

$$S^{eq} = h_1 2K_1 a_1 \frac{n_1^2}{R_1} + h_2 2K_2 a_2 \frac{n_2^2}{R_2} + h_3 b \left[ \frac{b}{2} - \eta_1 + \rho_n \left( \frac{\rho_n}{b} - 1 \right) \right] + 2h_4 b_1 \left\{ \frac{\rho_n}{b_1} - \frac{b_1}{2} - \eta_1 \right\}$$

(32)

Equation (32) represents the modified static moment of the equivalent cross-section (fig. 1.b) with respect to its neutral axis.

2.3.4. The stress in the flange

When analysing the stresses in the flanges due to bending the maximum moments \( M_x \) and \( M_t \) at the points of junction of the flanges and the web are determined as:

$$\begin{align*}
(M_x)_{x=a_1} &= D_1 \left( \frac{d^2 w_1}{dx^2} \right) = D_1 n_1 \frac{\Delta d\theta}{d\theta} \frac{4k_2}{V_1(k_1 a_1) V_3(k_1 a_1) + 4V_4^2(k_1 a_1)} V_1(k_1 a_1) V_3(k_1 a_1) + 4V_4^2(k_1 a_1) \\
(M_x)_{x=a_2} &= D_2 \left( \frac{d^2 w_2}{dx^2} \right) = D_2 n_2 \frac{\Delta d\theta}{d\theta} \frac{4k_2}{V_1(k_2 a_2) V_3(k_2 a_2) + 4V_4^2(k_2 a_2)} V_1(k_2 a_2) V_3(k_2 a_2) + 4V_4^2(k_2 a_2)
\end{align*}$$

(33)
The stresses in the middle layer of the inside and outside flanges respectively are:

\[
\sigma_{t0} = \frac{T_{t1}}{h_1} ;
\]

\[
\sigma_{t20} = \frac{T_{t2}}{h_2}
\] (34)

The stresses due to bending in the inside and outside flanges are:

\[
\sigma_{t1, b} = \frac{6M_{t1}}{h_1} \left( \frac{6(M_{x})_{x=a1}}{h_1} \right) ;
\]

\[
\sigma_{t2, b} = \frac{6M_{t2}}{h_2} \left( \frac{6(M_{x})_{x=a2}}{h_2} \right)
\] (35)

Substituting the quantity \( E \frac{\Delta d_6}{d_6} \) through equation (31) into equations (34) and (35) and solving it through equation (33), the total stresses in the outside and inside flanges are found to be:

\[
\left( \sigma_{t1x} \right)_{x=a1} = \frac{T_{t1}}{h_1} + \nu \left( \frac{6(M_{x})_{x=a1}}{h_1} \right) = \frac{M \eta_1}{S_{eqR1}} \left[ 1 + \nu K_{\sigma1} \right];
\]

\[
\left( \sigma_{t2x} \right)_{x=a2} = \frac{T_{t2}}{h_2} + \nu \left( \frac{6(M_{x})_{x=a2}}{h_2} \right) = \frac{M \eta_2}{S_{eqR2}} \left[ 1 + \nu K_{\sigma2} \right]
\] (36)

Where

\[
K_{\sigma1} = \sqrt{\frac{3}{1-\nu^2}} \left( \frac{\text{ch}(2k_1a_1) - \cos (2k_1a_1)}{2+\text{ch}(2k_1a_1) + \cos (2k_1a_1)} \right);
\]

\[
K_{\sigma2} = \sqrt{\frac{3}{1-\nu^2}} \left( \frac{\text{ch}(2k_2a_2) - \cos (2k_2a_2)}{2+\text{ch}(2k_2a_2) + \cos (2k_2a_2)} \right)
\] (37)

2.3.5. The stresses in the web

The total maximum tensile stress in the web is composed of the stress in the central layer of the web and the stress due to bending moment \( M_a \) at the junction point of the web and flange. The stresses in the central layer of the web are determined as:

\[
\sigma_{t, 0} = \frac{M (\rho_n - r)}{S_{eq, r}}
\] (38)

where \( r \) is the radius of curvature of the fibre considered. The stresses due to bending moment \( M_a \) are determined as:
The net maximum stress in a cross-section of the corner zone is determined as the sum of the stresses due to bending and due to tension:

\[ \sigma_{t.b} = \frac{Eh^3}{2(1-\nu^2)} \left( \frac{1}{r} \frac{d^2 w}{dr^2} + \nu \frac{d^2 w}{dr^2} \right) = \]

\[ = \frac{Mn_1 R_1^2 \left( 1 + \frac{h_2^2}{r^2} \right)}{R_2^2 - R_1^2} \left( \frac{h_1}{h_3} \right)^2 - \frac{1}{3} \left( \frac{h_1}{h_3} \right)^2 \frac{\cosh(2k_1 a_1) - \cos(2k_1 a_1)}{2 + \sinh(2k_1 a_1) + \sin(2k_1 a_1)} \]

Hence the total stress is found to be:

\[ \sigma_* = \frac{M}{S_{eq}} \left[ \frac{\rho_n}{r} - 1 + \left( \frac{3}{1-\nu^2} - 1 \right) \frac{\cosh(2k_1 a_1) - \cos(2k_1 a_1)}{2 + \sinh(2k_1 a_1) + \sin(2k_1 a_1)} \right] \]

\[ = \frac{n_1 R_1 \left( 1 + \frac{R_2^2}{r^2} \right)}{(R_2^2 - R_1^2)} \left( \frac{h_1}{h_3} \right)^2 \]

(39)

2.3.6. The net stress in the cross-section

The net maximum stress in a cross-section of the corner zone is determined as the sum of the stresses due to bending and due to tension:

\[ \sigma_{\text{max}} = \frac{M_i (n_i + \frac{1}{2} h_i)}{S_{eq_i} (R_1 - \frac{1}{2} h_i)} \left[ 1 + \nu K_{\sigma_i} \right] + \frac{N_i}{A_i} \]  \hspace{1cm} (41)

where \( M_i = F (a_c + \rho_{ci} \cos \varphi_i) \) and \( N_i = F \cos \varphi_i \), \( A_i \) is the cross-sectional area of the \( i \)-th section of the corner zone and \( M_i, n_i, S_{eq_i}, K_{\sigma_i} \) and \( N_i \) correspond to that cross-section. \( \rho_{ci} \) is the radius of curvature of the \( i \)-th centroidal line of the cross-section.

3. RESULTS AND CONCLUSIONS

fig 5

(figsize of page is 11)
1. Fig. 5 shows the relationship between the net maximum stress at the midpoint C (refer fig. 1.b) by different methods viz., conventional (method 1), the finite beam element (method 2) and thin wall-curved beam as a function of the angle $\phi$. The angle $\phi$ describes the position of the cross-section of the corner zone with respect to plane 1-1. It can be seen that the method 3 agrees better with measurements than the methods 1 and 2. This is due to the fact that the method 3 takes into account the deformation of the contour of the cross-sectional profile, whereas methods 1 and 2 neglect this effect.

2. Fig. 6 shows the distribution of the net maximum stresses for points lying along the entire length of the corner zone at a distance of 77 mm from the midpoint of the inside flange ($x/a = 0.6$). In view of the fact that the measuring points for $\phi = 0^\circ$ and $\phi = 7.5^\circ$ are located at 120 mm from the midpoint C (refer fig. 1.b) and for $\phi = 85^\circ$ at a distance of 77 mm, the measured stress values corresponding to $\phi = 0^\circ$ and $\phi = 7.5^\circ$ have been corrected for $x/a = 0.6$. 
3. Fig. 7 shows the distribution of the net maximum stress along the width of the inside flange for the cross-section A-A (refer fig. 1.b) as obtained by method 3. The maximum tensile stress attains the highest value at the midpoint of the inside flange and gradually decreases in the outward direction. Hence it may be concluded that in designing the corner zone, the application of a beam with a wide flange should be avoided as a considerable portion of the beam does not effectively contribute to the bending stiffness.

4. Fig. 8 shows the distribution of maximum stress (σ) along the central axis (Z-Z) over the entire height of the cross-section A-A (refer fig. 1.b).
5. Fig. 9 shows the relationship between the width reduction factor $K_1$ and dimensionless factor $a_1^2/R_1 h_1$ and similarly between the stress concentration factor $K_0$ and $a_2^2/R_1 h_1$.

From this it may be concluded that
a. with respect to strength it is not effective to choose
\[ a_1^2/R_1 h_1 > 1.8 \]
b. however, with respect to bending stiffness a virtual optimum is achieved for
\[ a_2^2/R_1 h_1 \approx 6 \]

In the latter case eq. (25) reduces to
\[ K_1 \approx \frac{1}{k_1 a_1} \quad (42) \]

REFERENCES